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PRELIMINARY

THERMAL PERFORMANCE OF STIRLING CYCLE CRYOCOOLERS: A COMPARISON OF JPL-TESTED COOLERS

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ABSTRACT

Spacecraft cryocoolers continue to attract interest in the space science community. To date, many Stirling cycle cryocoolers have been characterized by JPL in a number of performance areas including thermal performance. A number of factors arc used to evaluate the efficiency and cooling power of cryocoolers at cryogenic coldtip temperatures. A comparison of the measured thermal performance of several cryocoolers is presented.

The various cryocoolers tested by JPL include single compressor/single displacer, back-to-back dual compressor/dual displacer, back-to-back dual compressor/single displacer, and in-line single compressor/single displacer configurations. These cryocoolers were designed with various compressor and displacer sizes and strokes and were designed to operate at a drive frequency that is particular to each cooler. The design differences between various coolers complicates their comparison. This paper aims to evaluate the thermal performance data so that cooler-to-cooler comparisons can be made, As part of these comparisons, the paper explores the measured sensitivity of thermal performance to a wide range of operational variables. These include piston and displacer stroke, piston/displacer phase, fill pressure, heatsink temperature, and drive frequency. A multivariable plot format aids in the understanding of the complex interdependence of input power, coldtip temperature, coldtip heat load, specific power (efficiency), and the operational variable.

This paper provides the community with insight into the expected performance and limitations of state-of-the-art cryocoolers.

INTRODUCTION

Spacecraft cryocoolers continue to attract interest in the space science community. To date, many Stirling-cycle cryocoolers from manufacturers such as British Aerospace, Hughes, Lockheed, Stirling Technology Co., Sunpower Inc., Texas Instruments, and TRW have been characterized by JPL in a number of performance areas including thermal performance. The various cryocoolers include single compressor/single displacer, back-to-back dual compressor/dual displacer, back-to-back dual compressor/single displacer, in-line single compressor/single displacer configurations with either rotary or linear motors and some with passively driven displacers. These cryocoolers were designed with various compressor and displacer sizes and strokes and were designed to operate at a drive frequency that is particular to each cooler. The design differences between various coolers complicates their comparison.

This paper aims to evaluate the thermal performance data so that cooler-to-cooler comparisons can bc made. An overview of the thermal characterization procedure is provided to indicate the efforts required to obtain accurate characterization. The general sensitivity trends of thermal performance to piston and displacer stroke, piston/displacer phase, fill pressure, heatsink temperature, and drive frequency arc discussed using data taken from cryocoolcrs that best display the trends. Throughout this trend analysis a multivariable plot format is used to aid in the understanding of the complex interdependence of input power, coldtip temperature, coldtip heat load, specific power (efficiency), and the operational variable. To close the paper, several plots that include combined data sets from several coolers arc used to highlight performance and efficiency trends of the various coolers.

THERMAL PERFORMANCE CHARACTERIZATION PROCEDURE

The refrigeration performance measurements were conducted in the JPL thermal-vacuum test facility [1] that is used to simulate conditions in space and to provide a highly stable thermal test environment; the high level of environmental stability allows accurate repeatable measurements to illuminate subtle and important performance sensitivities. Each cooler component was attached to a copper flange, thermally isolated from the vacuum housing, to allow accurate control of the heatsink temperature using a fluid-loop heat exchanger. Heatsink temperatures and case temperatures of the cooler were monitored using a large number of thermocouples. The coldtip was outfitted with a cryo-diode to measure its temperature and a metal film resistor was used to apply a heatload. The coldtip was wrapped in several layers of aluminized Kapton MLI to reduce parasitic radiation heat load to negligible levels and the vacuum was maintained below 10^{-5} torr to avoid gaseous conduction effects.

In general, the coolers were driven using low distortion audio amplifiers with a sinusoidal voltage waveform". The power to the cooler components was monitored using high-quality true-RMS power meters. Because drive-cable ohmic losses arc also read by the power meters, cable ohmic losses were separately measured and were subtracted out in the final power data that are reported in the figures. Some coolers required DC power and others were supplied with their own linear or PWM drive electronics. Accurate measurement of the input power along with careful measurement of the drive motor resistance enables the determination of power factor, motor efficiency, and %Carnot COP for the coolers. These performance measures arc defined below in the final section of this paper.

Cryocoolers were tested according to a test matrix that includes studies of the sensitivity of thermal performance to compressor and displacer stroke, compressor/displacer phase, heatsink temperature, and drive frequency. In some cases, the fill pressure is also an accessible parameter and performance is measured at various pressures. Some coolers use a passive displacer that is driven by the pressure wave from the compressor which can preempt any study of compressor/displacer phase or displacer stroke. Other coolers operate at constant compressor stroke but allow for variation in the drive frequency. In these and other special cases, the test matrix is modified to accommodate the idiosyncrasies of the various coolers. Descriptions of the generally observed performance trends arc provided in the next section.

In addition to the test matrix described above for the thermal performance characterization, there arc other related aspects that arc investigated. Cooldown measurements arc made to determine the speed with which particular coolers achieve cryogenic temperatures and to provide a non-equilibrium temperature sweep of cooler operating parameters such as power factor and motor efficiency. Duc to the stability of the thermal environment, coldtip temperature stability can be evaluated over lengthy time scales. The parasitic heat conduction of the coldfinger is measured in a separate facility [2] to determine the heat load that would be imposed by a backup cooler that is not operational. This parasitic heat load can indicate a means to achieve greater heat lift from a particular cooler; since the cooler must cool its applied heat load in addition to the parasitic heat. These issues arc covered in more detail in the references [3,4,5,6,7]

GENERALLY OBSERVED PERFORMANCE TRENDS

As a result of the large number of coolers that have been characterized by JPL in a uniform fashion; a favorable position is generated that facilitates the observation of generally observed trends that appear despite the large differences in the design of the various machines. This section is dedicated to the discussion of these thermal performance trends. In each of the sensitivities discussed below, data were chosen from a particular cooler that was representative of the trends. It is the intention of the authors that these trends be used to facilitate the application and design of cryocoolers by providing some performance insights.

In all the sensitivity cases discussed below, the data arc plotted in a multivariable format that shows the interrelationship between the input power, the coldtip load, the specific power, the coldtip temperature, and the sensitivity variable. In all cases, the input power has been corrected for the losses in the leads that connect the power meter to the cooler. The curve fits through the data points arc quadratic polynomials and the intersection between the isotherms and the performance curves arc determined by quadratic curve fits to the coldtip load versus coldtip temperature data. The isotherms arc drawn as line segments that connect the interpolated points. The grid lines of specific power (input power/coldtip load) arc present on the plots to indicate whether a particular change in operating conditions yields better or worse performance with regard to overall refrigeration efficiency.

Compressor Stroke Fig. 1 demonstrated the sensitivity of the thermal performance to the compressor stroke. Curves arc shown for compressor strokes from 4.0 to 7.0 mm. Note that the displacer stroke, the compressor/displacer phase, the hcatsink temperature, the drive frequency and the fill pressure remain fixed for this sensitivity study. As expected, increasing the compressor stroke requires an increase in the input power. In general, the input power is weakl y dependent on the coldtip load and strongly dependent on the compressor stroke.

Isotherms arc shown for cold tip temperatures from 40 K to 150 K. Increasing the compressor stroke yields the same coldtip temperature at a greater coldtip load; therefore, more heat is lifted at higher strokes. Note that all the isotherms show this general trend because then lean to the right in the plot, toward higher coldtip loads. Now focus on the relationship between the isotherms and the specific power grid lines; an isotherm that follows a grid line demonstrates that no loss in refrigeration efficiency, at that coldtip temperature, is suffered as the stroke is increased, This trend is seen at the lower strokes on this plot. At the higher strokes, the isotherms arc more steeply sloped than the gridlines indicating that the efficiency decreases as the stroke is increased. For this particular cooler, the highest overall efficiency, at a given coldtip temperature, is yielded at the lower strokes. Higher strokes mean higher currents and higher motor 1OSSCS that result in lower efficiencies.

Displacer Stroke The sensitivity of the thermal performance to the displacer stroke is displayed in Fig. 2; once again, the other sensitivity variables remain fixed. Lines of constant displacer stroke, from 2.4 mm to 3.0 mm, arc parallel to one another over this range of coldtip loads. Increasing the displacer stroke increases the amount of required input power. However, it is clear from the isotherms that the highest efficiency is achieved at the highest displacer stroke with non-zero coldtip heat loads. The isotherms in this plot do not achieve a maximum in the efficiency as they did in the previous plot that depicted the affect of compressor stroke on the thermal performance. This indicates that it would be desirable to be able to increase the displacer stroke even further until the efficiency reached its maximum achievable value.

Compressor/Displacer Phase The next figure depicts the affect of compressor/displacer phase on the thermal performance. The lines of constant phase, shown in Fig. 3, extend from a phase of 45° to 75° and are all nearly parallel to one another. The parallel nature of the sensitivity curves is generally observed. As the phase is increased the input power increases; except for the measurements at 64° and at 70° phase, which overlap one another. Despite their overlap in input power, the coldtip temperature is not the same at both phases; note, that the isotherms make a sharp retreat to lower coldtip loads above the 64" phase curve. This indicates that there is a desired phase that results in the highest efficiency. At coldtip loads near 0.7 watts the desired phase is about 64° and at coldtip loads less than 0.3 watts the desired phase is about 55°. This trend in the optimum phase is generally observed, the higher the coldtip load, the higher the desired phase.

Fill Pressure Of all the sensitivity variables discussed in this section, the fill pressure is the one that has been the least available. In general, coolers that have been characterized by JPL are scaled units that do not allow access to the fill gas. Fig. 4 depicts data from a cooler that allowed the variation of the fill gas pressure. A set of three curves are shown that span a range of fill pressures from 180 psi to 240 psi; the standard operating pressure for this cooleris210 psi. Note, that the lines of constant fill pressure are nearly parallel to one another and the higher pressures require larger input power. In general, the isotherms are more steeply sloped than the specific power grid lines; indicating that the efficiency decreases with increased fill pressure. Above 80 K, the amount of useful refrigeration at a constant coldtip temperature increases with increased fill pressure. At coldtip temperatures below 70 K, no gain in refrigeration capacity is realized by increasing the fill pressure of this cooler, and the efficiency is seen to fall off sharply. Clearly there is a trade off between the amount of lifted heat and the efficiency of the cooler. Reducing the fill pressure yields higher efficiency but also adjusts the resonance of the cooler, adjusting the drive frequency may also be necessary to get the optimal performance at a particular fill pressure.

Heatsink Temperature Fig. 5 depicts the change in thermal performance as a result of a change in heatsink temperature. Two horizontal load-lines are presented for each of the two strokes (7.5 mm and 10 mm); one load line corresponds to a 20"C heatsink temperature, and the other to a 40"C heatsink temperature. Note that the two load lines for the same stroke lie nearly on top of one another; this implies that changing the heatsink temperature has a negligible effect on the input power required for a constant stroke and cold-tip load. This is also typical of many Stirling coolers.

Note that increasing the heat sink temperature from 20"C to 40"C causes the isotherms to shift to the left by about 5 to 7 K. As a result, decreasing the heatsink temperature decreases the input power required to refrigerate a particular coldtip load at a particular cold-tip temperature. Reducing the heatsink temperature increases the refrigeration performance at a particular coldtip temperature. Similarly, if the stroke is held constant, increasing the heatsink temperature by 20°C increases the cold-tip temperature by about 5 to 7 K for a fixed cold-tip load.

Drive Frequency Fig. 6 demonstrates the affect of drive frequency on refrigeration performance. Data are shown for drive frequencies of 50 Hz to 65 Hz in steps of 5 Hz. Note that specific power performance at a particular cold-tip temperature is a strong function of drive frequency; specific power is seen to degrade rapidly as the frequency deviates from the 60-Hz baseline value. When the drive frequency is adjusted from its design point, the efficiency of the cooler suffers due to off-resonance operation. It is worth noting, that these data are taken from a cooler that has a passive displacer; therefore, the affect of drive frequency on the thermal performance is stronger than it would bc in a driven displacer cooler. This is due to the fact that the displacer stroke is reduced when the drive frequency is adjusted from the design frequency.

COOLER COMPARISONS

To close the paper, several plots that include combined data sets from several coolers arc presented in this section to highlight performance and efficiency trends of the various coolers. This comparison serves to illuminate the tradeoffs that have been made in the various machines to attain their desired goals of low power consumption, low coldtip temperature, or rapid cooldown. To facilitate this study, performance curves from the various coolers have been selected for presentation on the same set of axes. Comparisons arc made in several performance measures that include load curves, multivariable format, %Carnot COP, %motor cfficicncy, and power factor. Once again, it is the intention of the authors that these comparisons be used to facilitate the application and design of cryocoolers by providing some insight into the necessary trade-offs.

Load Curves Fig. 7 depicts load curves measured for six different cooler configurations. Note, that all the curves have a concave down shape and exhibit an asymptotic or saturation phenomenon in the heat lift. In general, a trade off is seen between achievement of low temperatures and high lift at higher temperatures. A notable exception is shown in the data indicated with open triangles; this is a cooler that requires high input power and is designed for short-life and rapid cooldown. The other coolers are designed for longer life and/or lower input power. It is clear from this plot that it is possible to lift substantial heat loads at low cryogenic temperatures; however, no indication of required input power or lifetime can be derived from this presentation. The cryocoolcr customer is often interested in cooling a particular load to a particular temperature with less than some defined amount of input power; this plot does not provide all the necessary information.

Multivariable Plot Fig. 8 depicts data from five cryocoolers that span nearly an order of magnitude in their required input power. Note that despite the extreme differences in the cryocooler designs, this multivariable format is capable of yielding a nice presentation. Isotherms arc shown for coldtip temperatures ranging from 40 K to 130 K in steps of 10 K. These isotherms immediately indicate the coldtip loads that the various coolers can maintain at a particular coldtip temperature. The higher powered cooler lifts more heat, more efficiently, at the higher temperatures than the lower powered coolers. The intermediate powered coolers achieve the lower temperatures with the highest efficiency.

A dashed box is outlined on the plot to show a possible customer requirement that 1.5 watts of load be cooled with less than 70 watts of input power. By looking at the intersection of the isotherms with the performance curves relative to this box, the customer can immediately determine the temperature to which their load can be cooled with the available coolers. For example, the upper cooler could refrigerate more than 5 watts of load to 100 K with the available input power; whereas the second cooler from the top could cool a load of 1.5 watts to 80 K. This type of presentation facilitates trade-offs/comparisons by the cooler user. This also brings up another point of comparison regarding the customer rcquircmcnts, throttling of the cooler. In the example dcscribcd, the upper cooler could refrigerate 5 watts to 100 K with the available power, but the customer has a minimum heat load requirement of 1.5 watts. Reduction of the stroke will consume less power; a smaller coldtip load will enable the cooler to cool to a lower temperature often with higher efficiency. By first comparing different coolers on the same plot and then considering throttling of larger coolers with cooler specific plots, such as Fig. 1, the customer can determine whether it makes more sense to usc a throttled-back larger cooler or to usc a smaller cooler.

%Carnot COP Compressor/displacer thermodynamic COP, is a measure of the ability of the cooler to convert work done on the gas into net cold-tip cooling power, it is defined in this paper as:

$$COP_{td} = \frac{P_{ct}}{P_{in} - I^2R}$$

Where P_{ct} represents the coldtip load, Pin represents the input power minus the lead losses, and R represents the coil resistance of the drive motor. In this expression the work done on the gas is approximated as the input electrical power minus the drive-motor i^2R losses. An important figure-of-merit for cryocoolers is the thermodynamic COP of the refrigerator expressed as a percentage of the ideal Carnot COP; this %Carnot COP is thus defined as

$$\%\text{Carnot COP} = 100 * \frac{COP_{td}}{\text{Ideal Carnot COP}}$$

$$\text{Where: Ideal Carnot COP} = \frac{T_{ct}}{T_{hs} - T_{ct}}$$

and T_{ct} and T_{hs} represent the coldtip and heatsink temperatures respectively.

Fig. 9, shows the measured %Carnot as a function of coldtip temperature for a set of 5 cryocoolers. All curves have a concave down shape and peak at some maximum value. The peaks range from about 12% to 22% for the coolers considered here. Note, that a cooler that is designed to achieve a colder no-load temperature has a lower maximum than a cooler that is designed to achieve a warmer no-load temperature. Clearly there is a design trade-off that yields this type of performance difference.

%Motor Efficiency and Power Factor Cooler drive motors consume power in three principal ways: 1) by doing useful work on the applied load, 2) by dissipating i^2R losses in the drive coil, and 3) by doing work to overcome various internal frictional forces impeding the motor motion. Frictional forces include windage, mechanical friction, and eddy current forces. Because i^2R losses are generally the dominant loss term in a good motor, cooler motor efficiency is defined in this report as

$$\%\text{Motor Efficiency} = 100* \frac{P_{in} - I^2R}{P_{in}}$$

where P_{in} represents the input power corrected for the lead losses. There are four principal means of minimizing i^2R losses: 1) maximize the magnetic flux density to minimize the current required to generate a given drive force, 2) minimize the coil resistance for a given number of coil turns, 3) minimize the operating temperature of the coil and magnet, and 4) minimize the capacitive or inductive circulating currents that contribute to i^2R losses, but do no useful work. Eliminating the circulating currents is the same as requiring that the motor have a near unity power factor, where power factor is defined as the cosine of the phase angle between the input drive voltage and the input drive current. The power factor is also defined as:

$$\text{power 'actor} = \frac{P}{V * I}$$

where P represents the raw measured power that includes the lead losses and V and I represent the rms drive voltage and drive current respectively.

A primary cause of non-unity power factor is the presence of compressor drive forces that are not in phase with the compressor velocity. The mechanical-spring and gas-spring forces that restrain the piston movement, and the inertial forces that are required to accelerate the piston, arc opposite in sign and 90° out of phase with the velocity; therefore, minimizing the sum of these forces yields a higher power factor, This condition of equal and opposite spring and inertial forces is known as the resonant condition for a cooler drive system. Achieving resonance at the cooler drive frequency is thus important to maximizing cooler motor efficiency.

The coldtip temperature is important to motor efficiency because it generally influences the cooler resonant frequency, and thus the power factor, and the total motor input power; the input power in-turn influences the current level and the coil temperature. Fig. 10 shows the sensitivity of the measured motor efficiency, of 5 cryocoolers, to the coldtip temperature. Generally, the motor efficiency drops off from a high value near 160 K to a lower value at the lowest temperatures achieved by a particular cooler; however, the motor efficiency variation of a particular cooler is fairly small. The cooler that exhibits efficiency near 50% is exceptional to the others compared here, since it is a very low power cooler and dots not suffer a large power loss as a result of this lower motor efficiency. It is preferable to have peak motor efficiency at the temperature that is specific to an application. For example, the cooler data plotted as open diamonds has a modest peak near 80 K. It is also desirable to attain the highest motor efficiency possible at a particular operating point. From this set of data it is clear that efficiency higher than 75% at cryogenic temperatures is an achievable mark.

A primary contributor to the loss of motor efficiency of a particular cooler is a drop in power factor. Fig. 11 shows the measured power factor as a function of coldtip temperature for 5 cryocoolers. The variation in the power factor of the various coolers is distinct to each cooler; some gently rise toward lower coldtip temperatures, whereas others reach a peak at a particular coldtip temperature. Ideally a designer would want to attain the largest power factor at the desired operating point for the cooler. If the power factor is a gentle function of the coldtip temperature, then the cooler would have a wider desired operating band. Coolers that achieve a peak in the power factor arc likely fine tuned with respect to their mechanical dynamics for that particular coldtip temperature.

CONCLUSIONS

Due to the experience that JPL has acquired through the characterization of many cryocoolers under uniform conditions, the acquired data from the coolers can be used to arrive at several general performance observations. The thermal performance measurements presented above, for several cryocoolers, have been used as examples to aid in the description of the generally observed trends that their performance follows. Despite the fact that these cryocoolers were designed with various compressor and displacer sizes and strokes and were designed to operate at a drive frequency that is particular to each cooler, the general trends were usually observed. The discussion of these general trends was facilitated by the usc of a multivariable plot format that reveals the complex interdependence of input power, coldtip temperature, coldtip heat load, specific power (efficiency), and the sensitivity variable.

Another advantage to characterizing many cryocoolers under uniform conditions is that the acquired data lend themselves to sensible comparison. As a result, several plots that include combined data sets from several coolers were used to highlight performance and efficiency trends of the various coolers. Through these comparisons, and through observations of general trends, the obtained results should be useful to the cryocooler design community and to the cryocooler customers.

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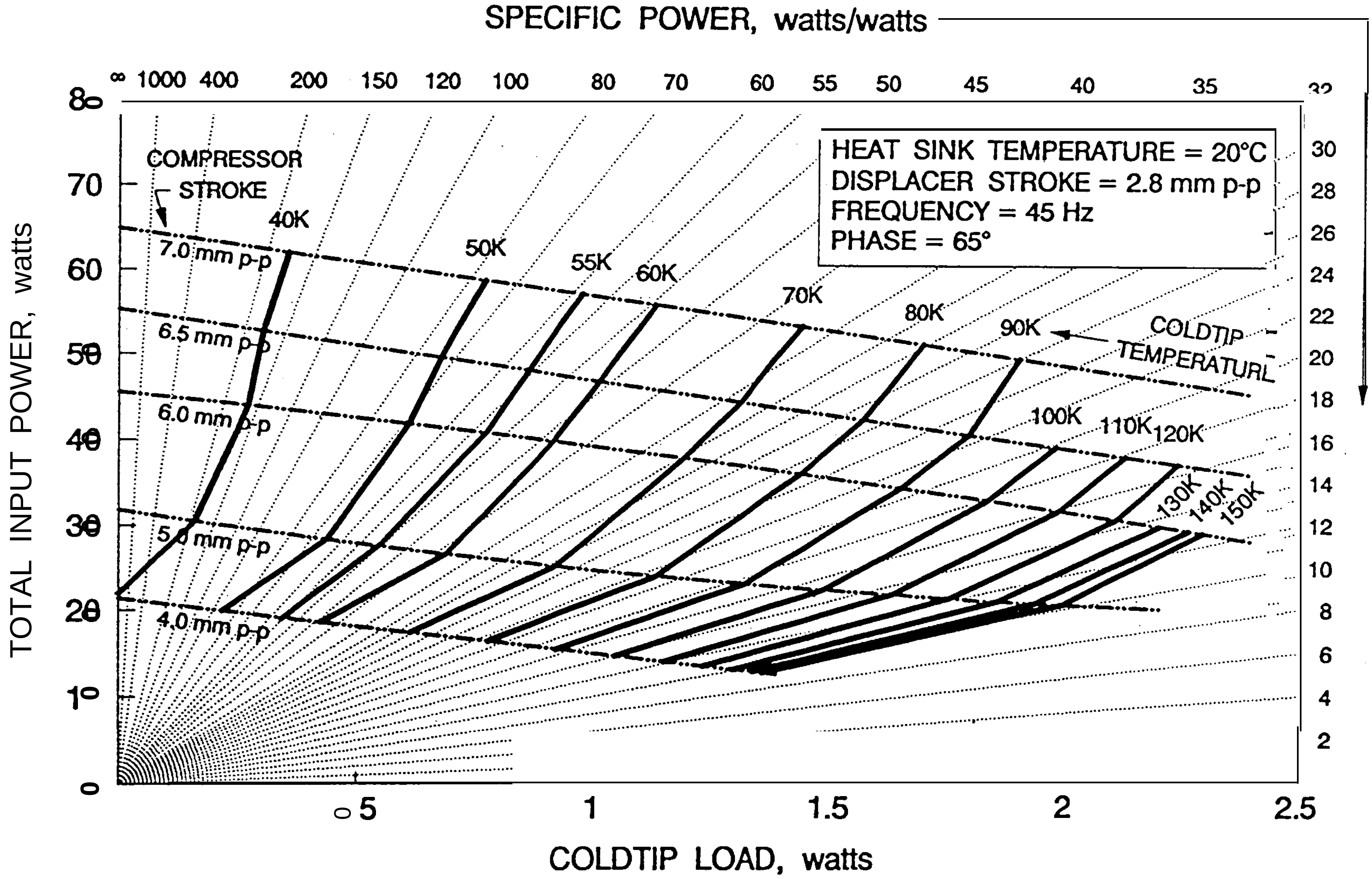
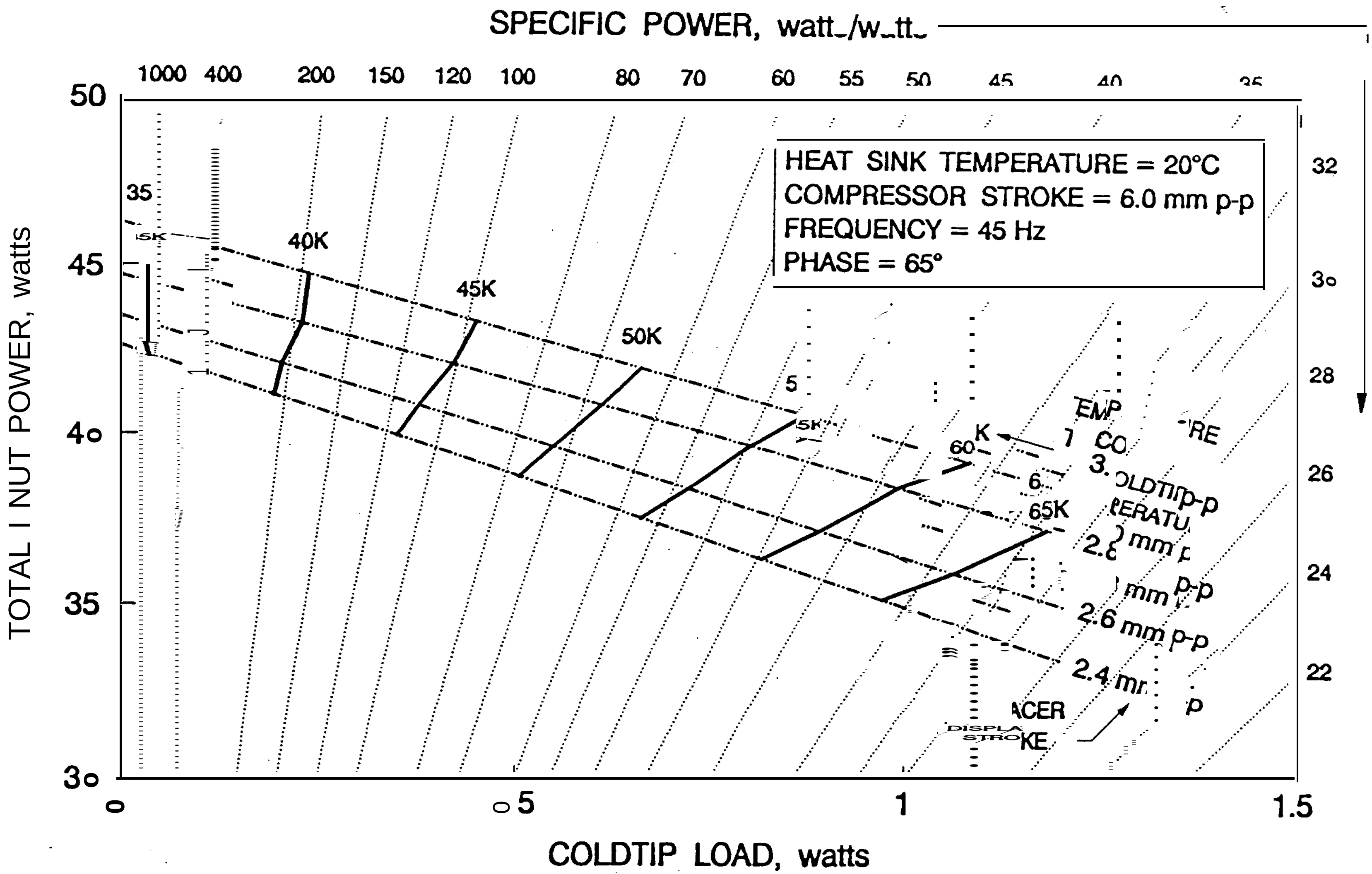
FIGURES

1 Compressor stroke sensitivity
2 Displacer stroke sensitivity
3 Compressor/Displacer phase sensitivity
4 Fill pressure sensitivity
5 Heatsink temperature sensitivity
6 Drive Frequency sensitivity
7 Load curves at nominal operation of various coolers
8 Multivariable plot at nominal operation of various coolers
9 %Carnot COP curves at nominal operation of various coolers
10 %Motor efficiency curves at nominal operation of various coolers
11 %Power factor curves at nominal operation of various coolers

BAe 50K TO 80K CRYOCOOLER
SENSITIVITY TxERMAL OF PERFORMANCE TO COMPRESSOR STROKE



BAe 50K TO 80K CRYOCOOLER
SENS T VITY OF THERMAL PERFORMANCE TO DISPLACER STROKE

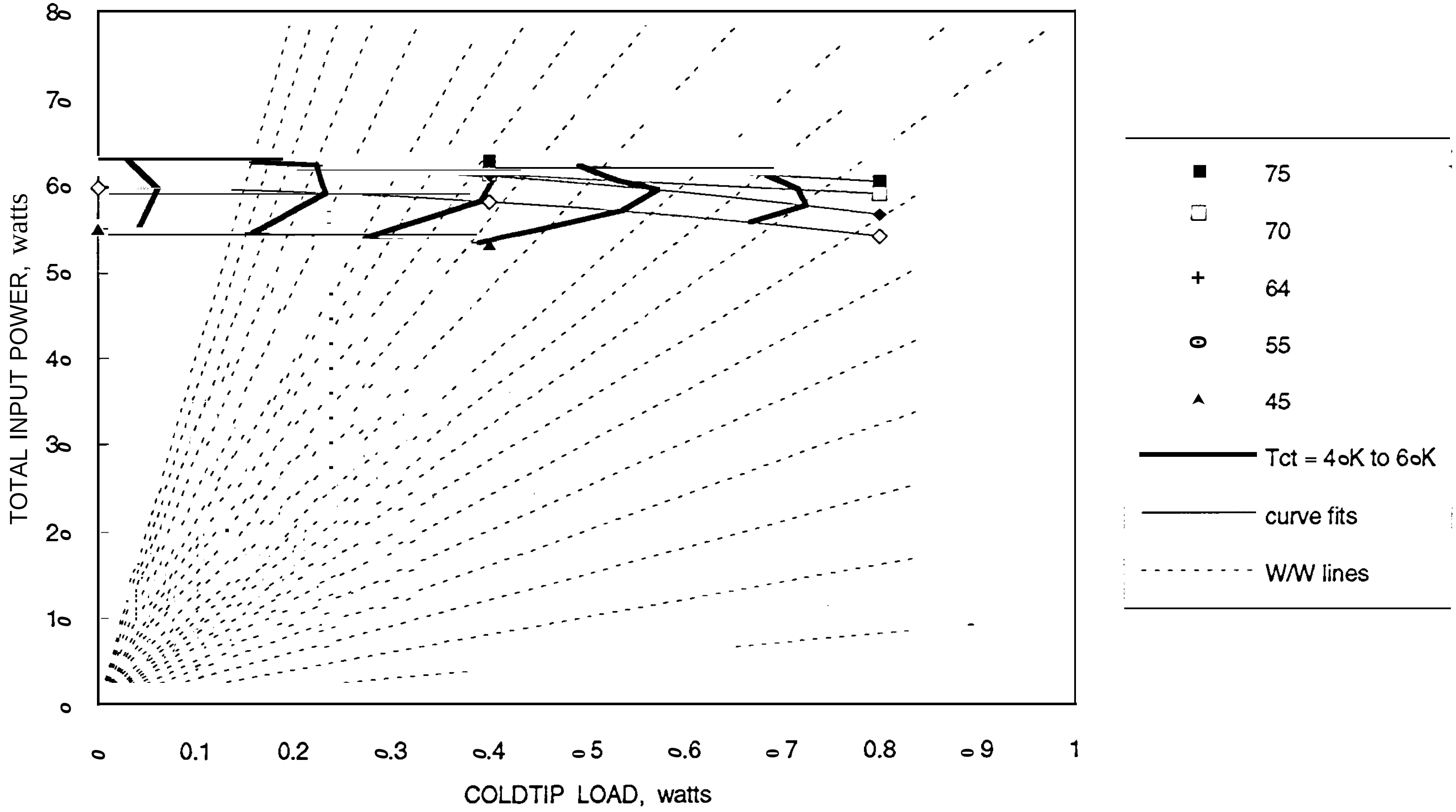


ESA_41.WK3 Fig. 2

ESA_41

BAe 55K DM CRYOCOOLER
SENSITIVITY OF THERMAL PERFORMANCE TO COMP/DISP PHASE

HEAT SINK TEMPERATURE = 20C; COMP STRK = 6.6 mm; DISP STRK = 3.0 mm

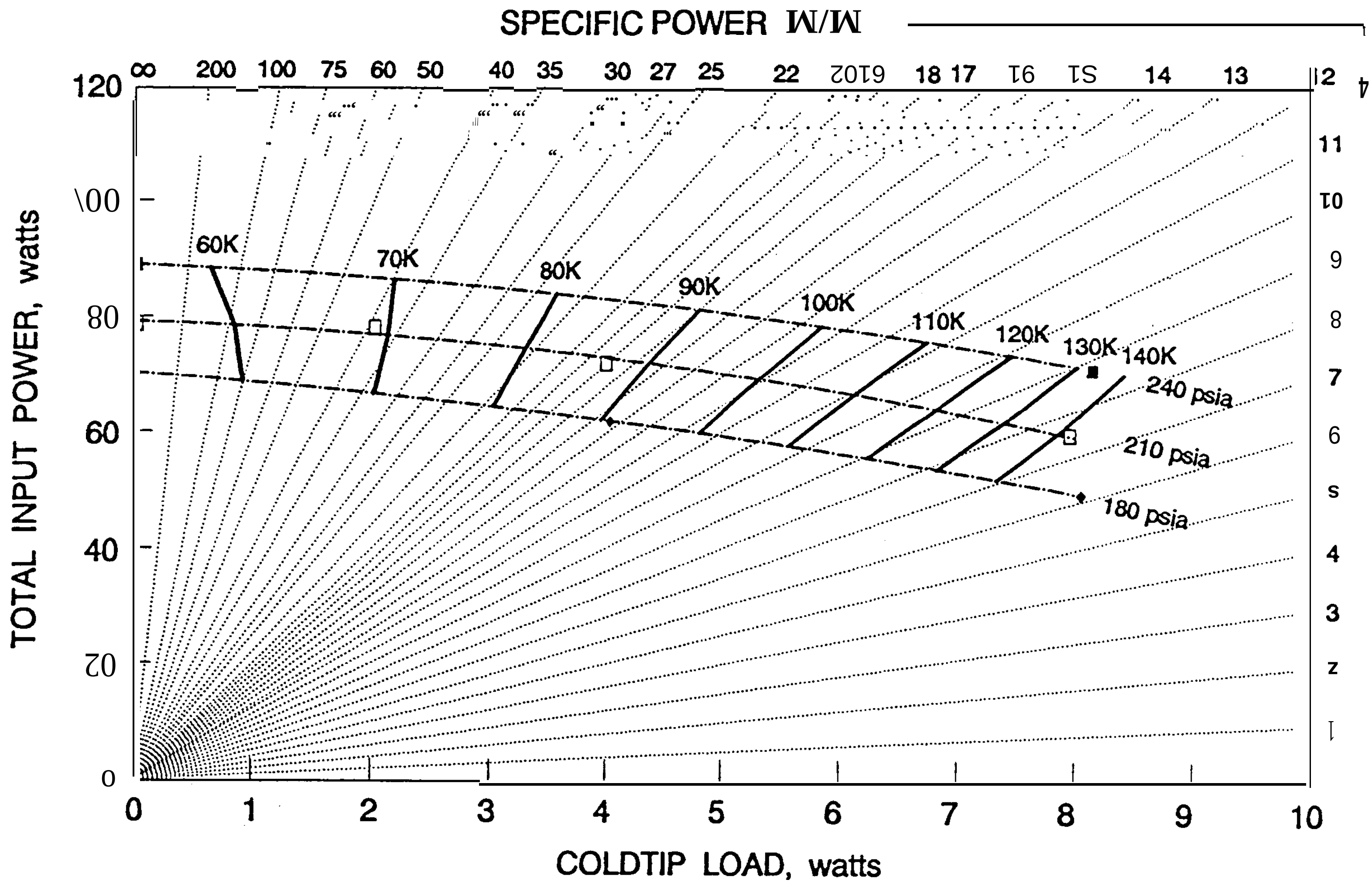


G. Smedley Fig. 3 4 19 94 [BAe55K_phsross.XLW]"Ross"-plot

STIRLING A901ONH331 80K COOLER

SENSITIVITY OF THERMAL PERFORMAN CE TO FILL PRESSURE

HEAT SINK = 20°C; DISPLACER STROKE = CONSTANT; COMPRESSOR STROKE = 10 mm



2-1

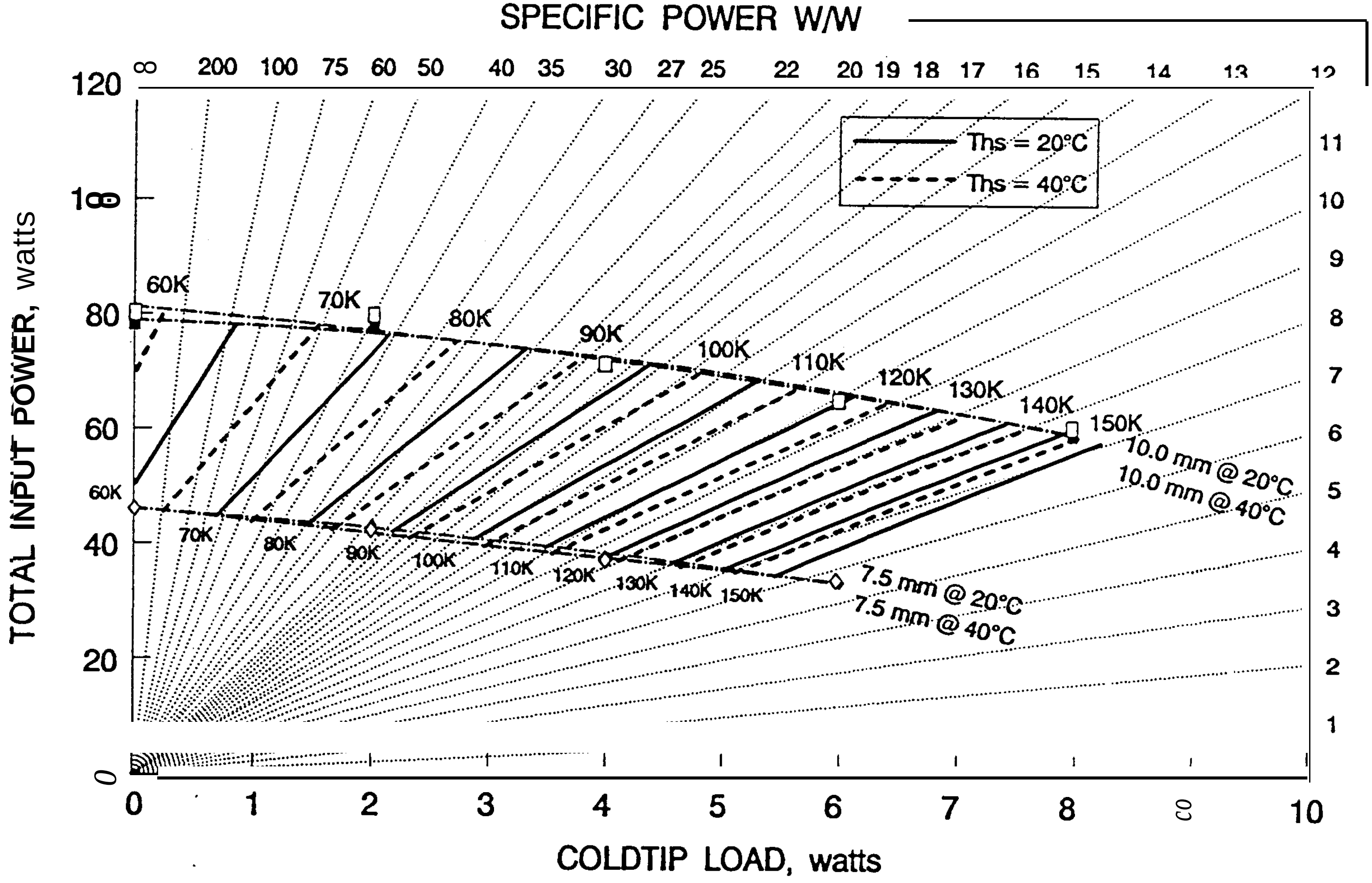
ROSS_PRESR.XLW ROSS_PLOT

Fig. 4

STC_082

STIRLING TECHNOLOGY COMPANY 80K COOLER SENSIT VITY OF THERMAL PERFORMANCE TO HEAT SINK TEMPERATURE

DISPLACER STROKE = CONSTANT; FILL PRESSURE = 2 0 psi

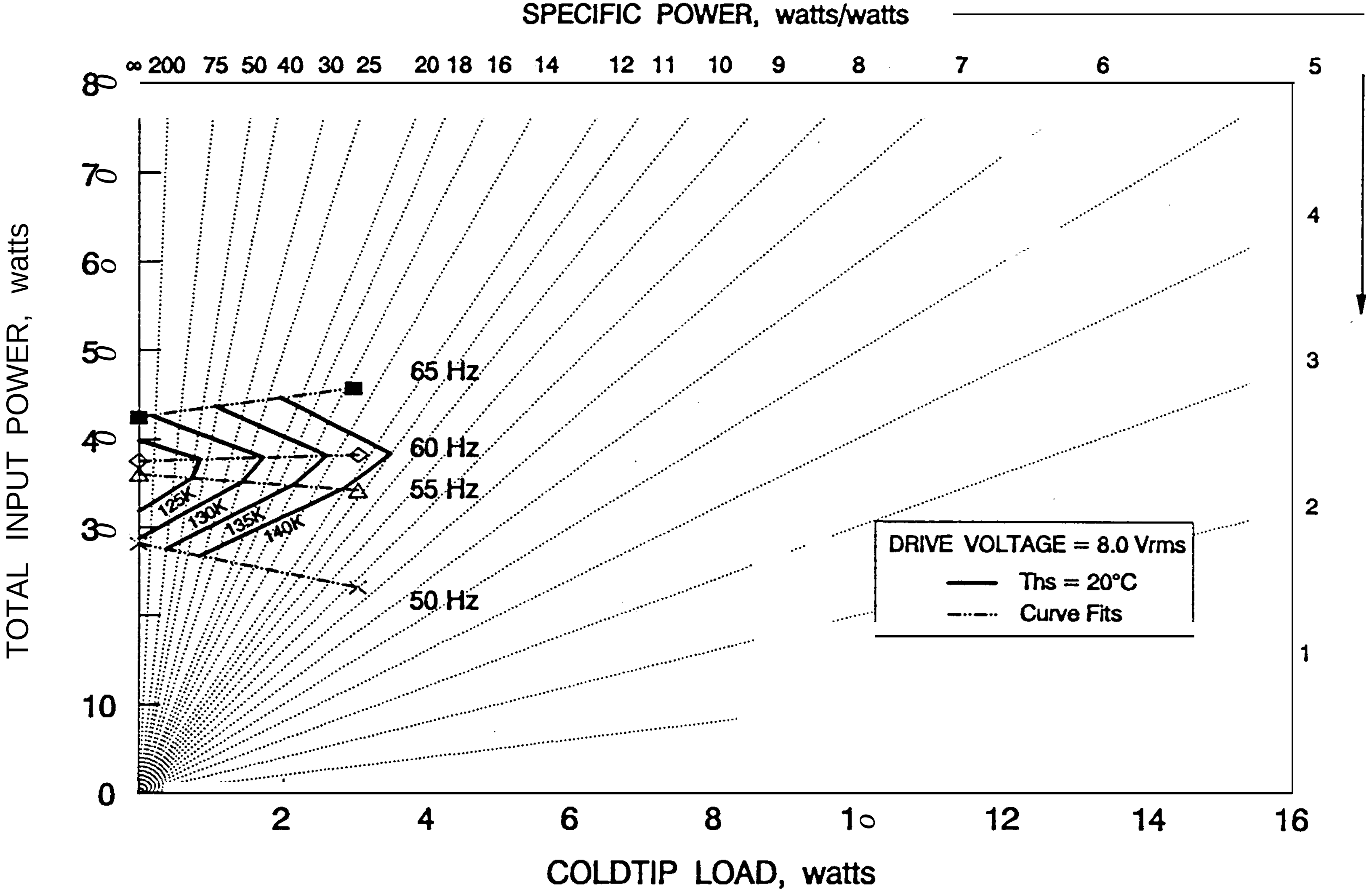


ROSS COMP.XLW ROSS_PLOT

Fig. 5

STC 080

SUNPOWER 40K COOLER
SENSITIVITY OF THERMAL PERFORMANCE TO DRIVE FREQUENCY

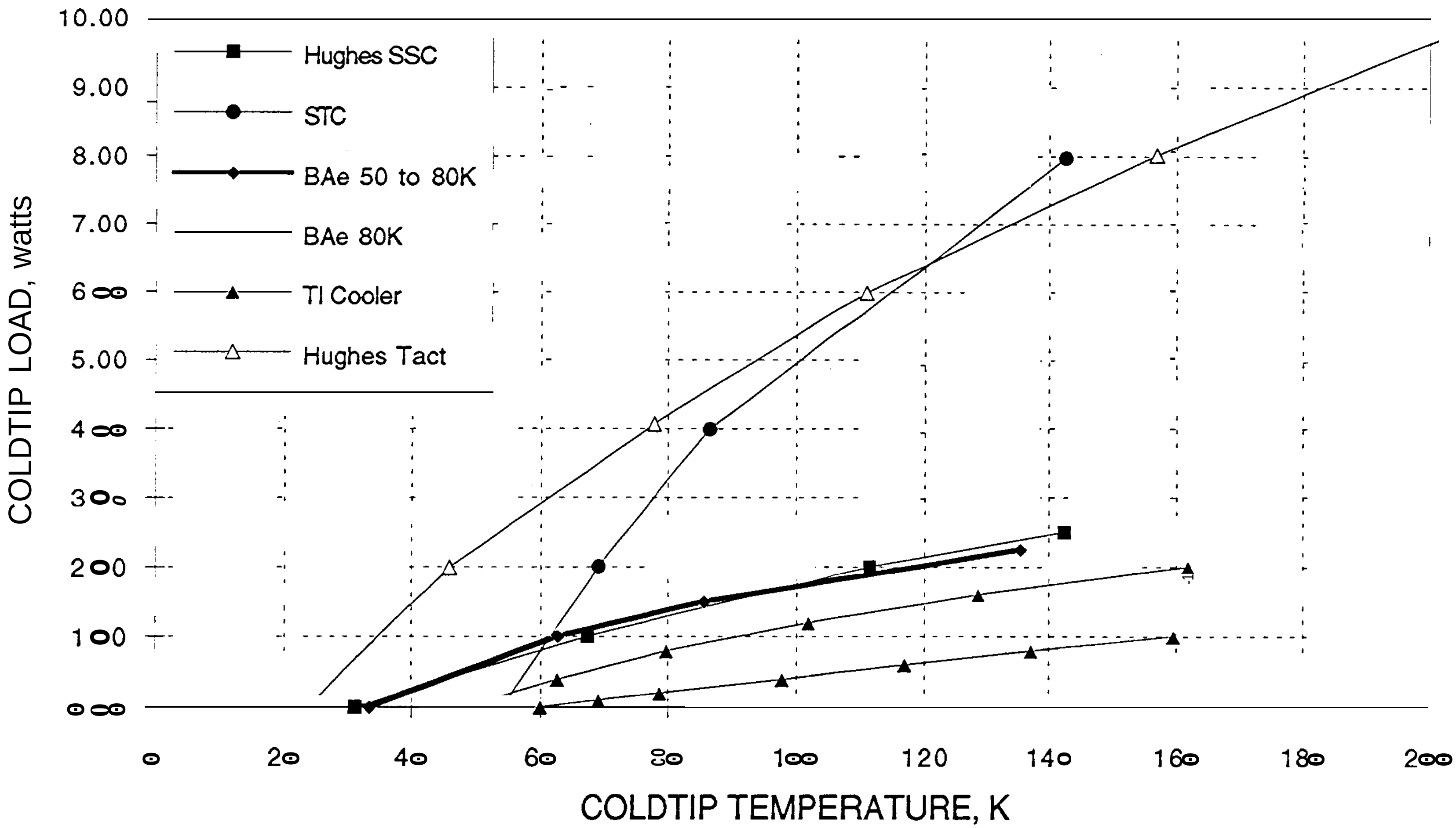


REQROSS WK3

Fig. 6

SUN_39

LOAD CURVE COMPARISON OF THERMAL PERFORMANCE
AT NOMINAL OPERATING CONDITIONS



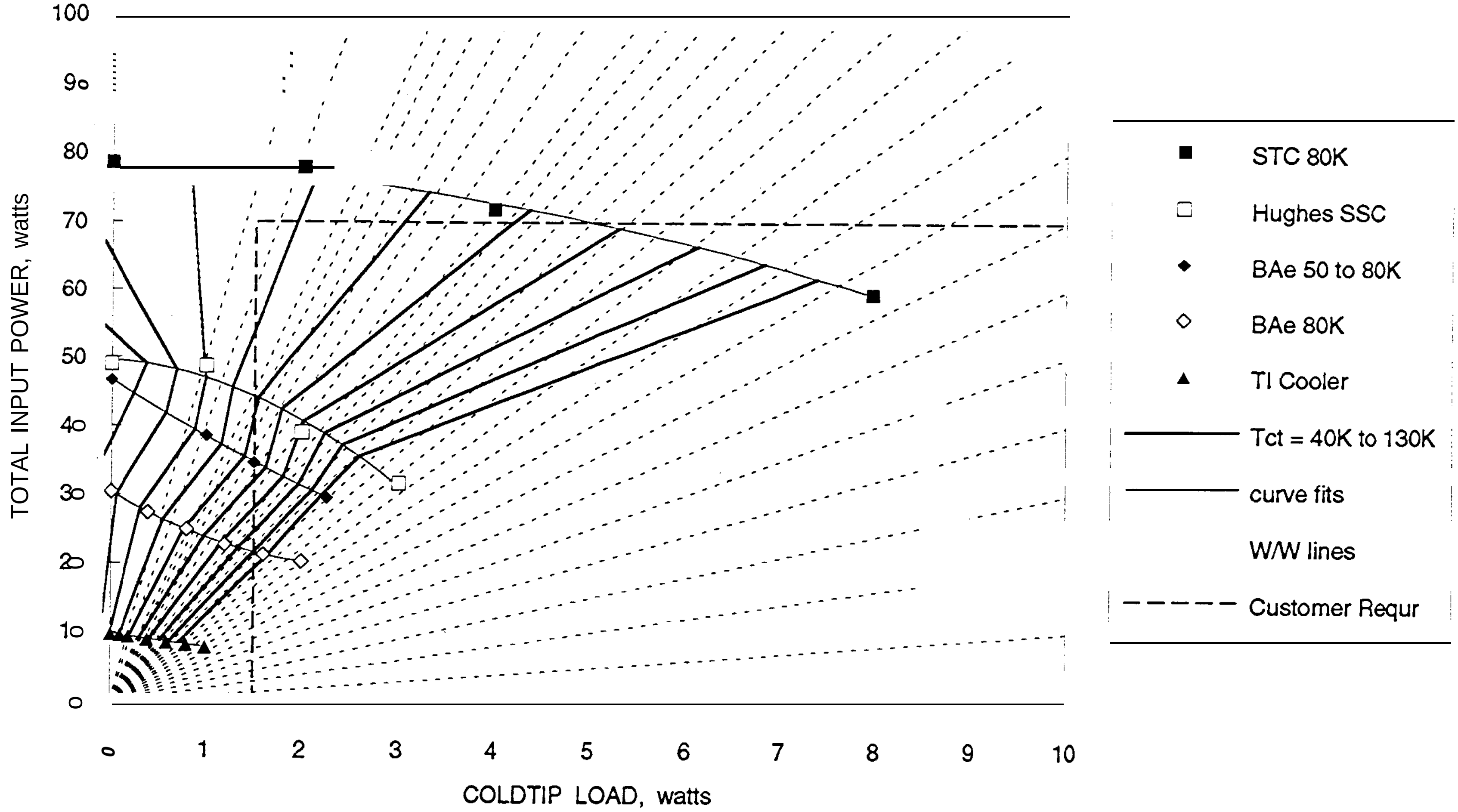
G ed

Fig. 7

4/13/94

[dimn.xlw]Load_Tct_nom

MULTIPARAMETER COMPARISON OF THERMAL PERFORMANCE AT NOMINAL OPERATING CONDITIONS



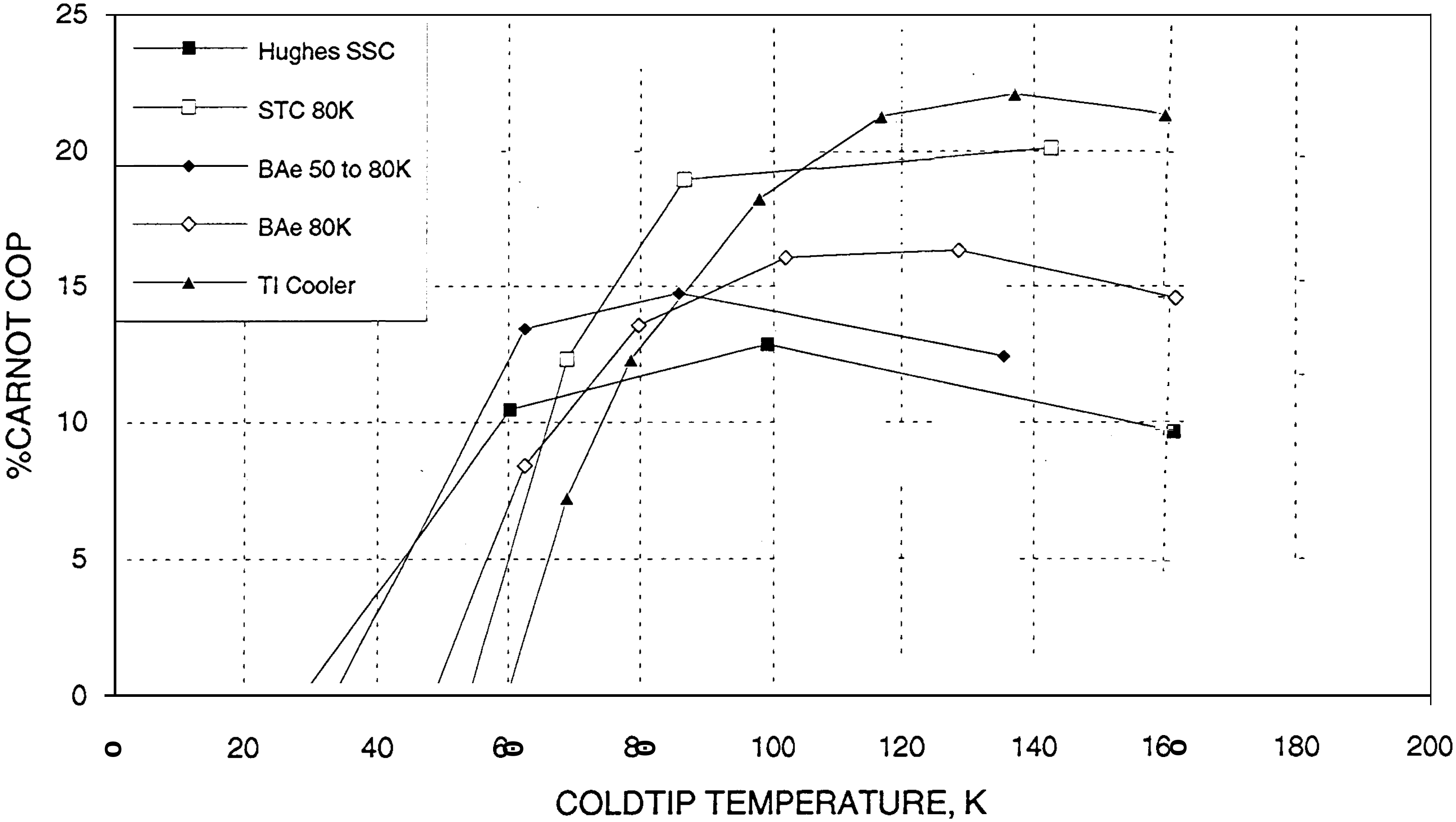
G. S ed

Fig. 8

4/15 94

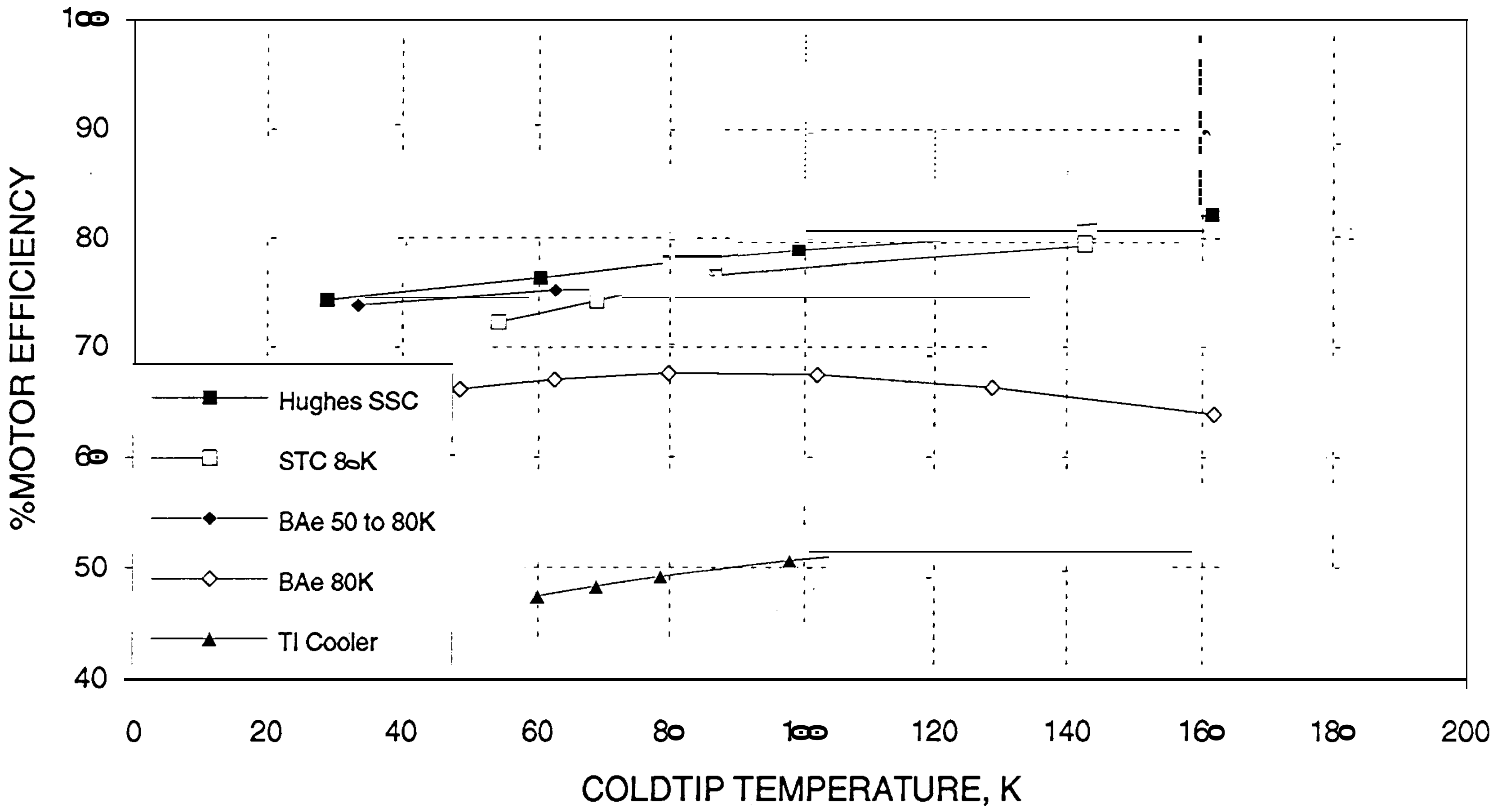
[Nominal_multi.xlw]"Ross"-plot

%CARNOT COP COMPARISON AT NOMINAL OPERATING CONDITIONS
SENSITIVITY TO COLDTIP TEMPERATURE



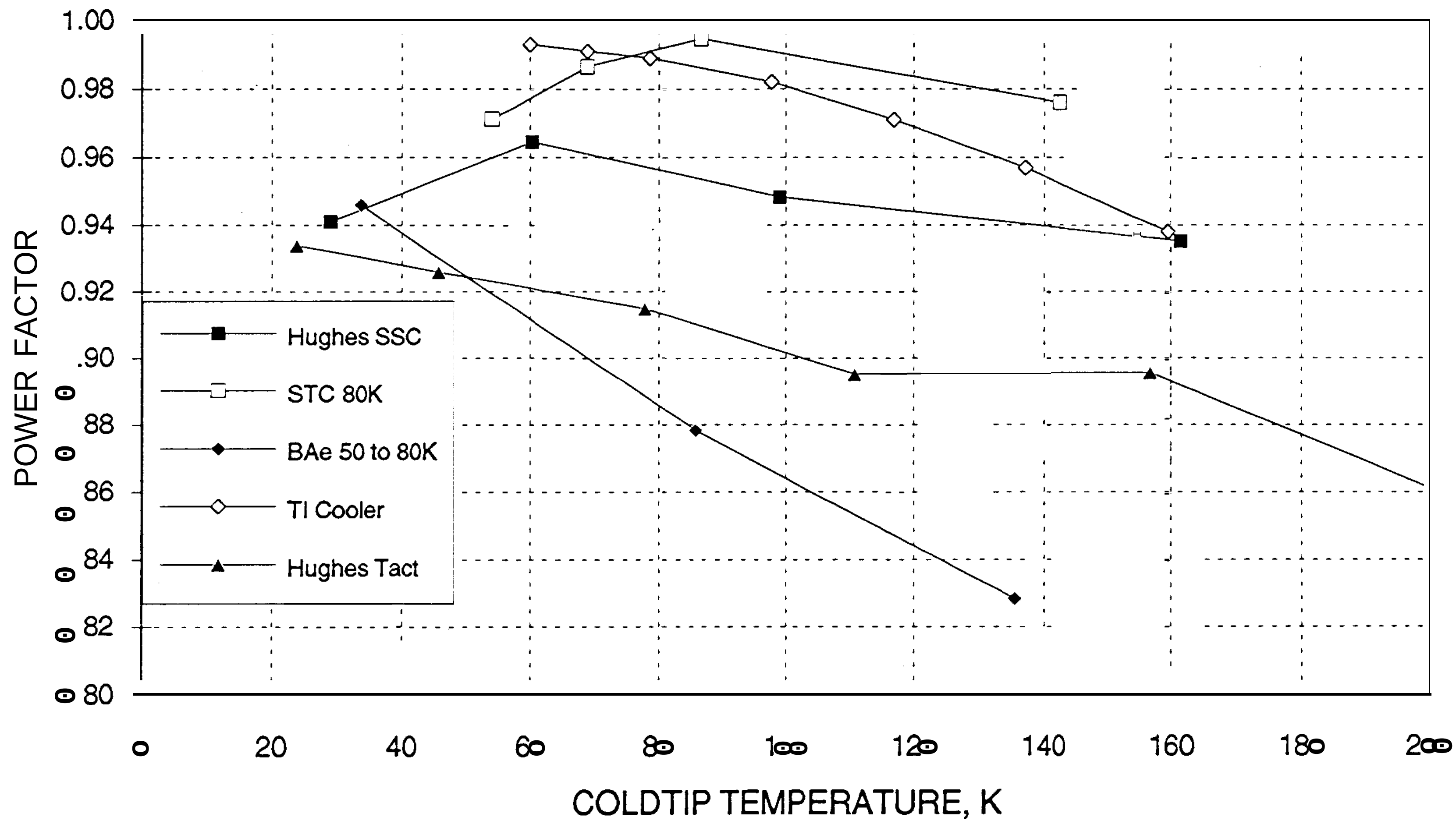
G. Smedley Fig. 9 4/14/94 [dimn.xlw]%COP_

%MOTOR EFFICIENCY COMPARISON AT NOMINAL OPERATING CONDITIONS
SENSITIVITY TO COLDTIP TEMPERATURE



G. Smedley Fig. 10 4/14/94 [dimn.xlw]%mtr_nom

POWER FACTOR COMPARISON AT NOMINAL OPERATING CONDITIONS SENSITIVITY TO COLDTIP TEMPERATURE



G ed *Fig 11* 4/13/94 powfa m